LEWIS GRANT
IN-44-CR
69259
P. 17

LOSS TERMS IN FREE-PISTON STIRLING ENGINE MODELS

Lloyd B. Gordon*
Space Power Institute
Auburn University
Auburn, Alabama

January 25, 1992

(NASA-CR-189840) LOSS TERMS IN FREE-PISTON STIRLING ENGINE MODELS Final Technical Report (Auburn Univ.) 17 p CSCL 10B

N92-17225
Unclas
G3/44 0069259

Prepared for
NASA Lewis Research Center

* Currently at the University of Texas at Arlington,
Department of Electrical Engineering

CONTENTS

LOSS TERMS IN FREE-PISTON STIRLING ENGINE MODELS

Lloyd B. Gordon

ABSTRACT

Various models for free-piston Stirling engines are reviewed. Initial models were developed primarily for design purposes and to predict operating parameters, especially efficiency. More recently, however, such models have been used to predict engine stability. Free-piston Stirling engines have no kinematic constraints and stability may not only be sensitive to the load, but also to various nonlinear loss and spring constants. This report reviews the present understanding of various loss mechanisms for free-piston Stirling engines and discusses how they have been incorporated into engine models.

1. INTRODUCTION

1.1 The Free-Piston Stirling Engine

A Stirling engine is a closed cycle machine in which heat input is converted into mechanical work through the alternate expansion and compression of a fluid (for this report, a gas). There are four discrete phases in an ideal Stirling engine thermodynamic cycle: isothermal compression with simultaneous heat rejection, constant volume regenerative heat addition, isothermal expansion with simultaneous heat addition, and constant volume regenerative heat removal. There are typically five separate spaces for the working fluid of a Stirling engine: expansion space, heater, regenerator, cooler, and compression space. In addition, there may be one or more gas regions which act as gas springs. The heat supplied during expansion is transferred to the gas through the heater. The cooler is a heat exchanger located between the regenerator and the compression space and removes heat from the working gas so that the compression process approaches an isothermal condition. A regenerator is a matrix of material that absorbs heat from the working gas as it moves from the expansion space to the compression space, stores the heat and releases it to the gas as it moves from the compression space back to the expansion space. A typical kinematic Stirling engine has one or more pistons constrained to each other and to the output shaft by levers, cranks, cams, and connecting rods.

A free-piston Stirling engine (FPSE) is a thermally driven mechanical oscillator that operates on a Stirling engine cycle. Unlike a kinematic Stirling engine the FPSE has unconstrained (by levers, cams, etc.) oscillating components. There are no mechanical linkages coupling the pistons or displacers. The FPSE engine typically has three basic components: a cylinder, a displacer and a power piston. The expansion space is between the cylinder and the displacer, the compression space is between the displacer and the power piston, and a bounce volume (which acts as a gas spring) is between the power piston and the cylinder. There are three types of forces that determine the engine dynamics: the force on components due to displacement of components (i.e., "spring" forces); the force on components due to velocity of components (i.e., damping forces); and the force on masses due to acceleration. Forces due to displacement and velocity may be determined from the thermodynamic properties of the cycle and the engine geometry. If the working fluid is assumed to obey the perfect gas law for adiabatic expansions and compressions, the force due to displacement will be proportional to the relative displacement and is analogous to a force in a linear spring. The spring action is due to the interaction between the pressurized fluid and the engine piston. The force due to the velocity is dissipative in nature and is proportional to the relative velocity. It is analogous to force in a linear damping.

The free-piston Stirling engine is a dynamic, resonant, system operating at more or less a constant frequency (at equilibrium) and is typically self-starting. When the expansion space is heated up from cold the system requires only a slight, random, perturbation to set it in motion. This natural self-starting capability of the free-piston Stirling engine is very important and differs from kinematic drive machines. Another major advantage to an FPSE is that there are no major side forces exerted by the reciprocating elements against the cylinder wall (due to the lack of any connecting rods). Elimination of the piston side-force reduces wear so that gas lubricated pistons can be used. Since the piston lubricant is also the working medium contamination is also eliminated (such as an oil lubricant would cause). The FPSE is also simple, low cost, with a long lifetime, as compared to kinematic Stirling engines.

The free-piston Stirling engine is essentially a vibrating system with a driver, a load, and losses. The system includes "spring" forces which act to restore the mass (piston or displacer) to a static equilibrium position, damping forces, input forces, and load forces. When a mass is displaced a distance x from static equilibrium it experiences a force proportional to x by a constant ($F = Kx$). This force causes the mass to move back toward the original equilibrium position. When the mass reaches the original equilibrium it will continue to pass (due to inertia) compressing the gas spring and experiencing a negative spring force

acting in the opposite direction. The oscillations occur at a characteristic frequency known as the natural frequency. In an FPSE the spring forces acting on the piston and displacer arise from gas springs supporting these reciprocating elements.

The gas pressure forces arise as a result of the cyclic variation of the working fluid pressure during the operation of the engine. Motion of the piston increases and decreases the total volume of the working space thereby creating a cyclic pressure variation. Energy is added to the system by the pressure increase in the expansion space resulting from the thermal input.

All vibrating systems experience some friction or resistance termed damping acting to slow down the motion and cause the oscillation to die away. Damping can arise from a variety of causes such as air damping, fluid friction, Coulomb dry friction, magnetic damping, or internal hysteresis. Damping forces always resist the motion of the mass and typically viscous damping is assumed where the resisting forces are proportional to the velocity. This results when the damping force is due to the viscous resistance in a fluid medium as in a ideal dashpot. The dashpot is characterized by a proportionality constant "C", called the coefficient of viscous damping, and the total damping force is represented by $C\dot{x}$, where $\dot{x}$ is the velocity of the mass.

Due to the absence of a kinematic drive the variables that characterize the engine dynamic response are the oscillation amplitudes of the moving elements, the frequency, and the phase angle. The motion of the mass can be described by Newton's Second Law of motion $\Sigma F = Ma$ where ΣF is the summation of all forces acting on the mass and a is the acceleration of the mass ($\ddot{x}$). Thus, Newton's law gives the equation of motion for the mass:

$$M\ddot{x} + D\dot{x} + Kx = \Sigma F_p + \Sigma F_e$$

Thus, in an FPSE the forces include the damping forces (characterized by D's), spring forces (characterized by K's), working fluid pressure forces (F_p's, which includes the thermal driving force), and external forces, primarily the load (F_e's). Often the gas pressure force terms are incorporated into the spring constant terms (K's) and the load damping terms are incorporated into the damping terms (D's). Since a FPSE typically has two moving elements (the piston and the displacer) linear engine dynamics can be expressed in terms of two second order differential equations with constant coefficients.

1.2 Stability and damping

The performance of an FPSE can be characterized by frequency, phase (between the piston and the displacer), power out, efficiency, and stability. Various modes of instabilities can occur in an FPSE, including:

- dropout - a condition in which amplitudes of oscillation decrease until engine oscillations cease
- blowup - a condition in which oscillations increase until limited by displacer or piston collisions
- hunting - a condition is which the amplitude and/or frequency periodically increase and decrease

Typically, the input energy source and physical engine characteristics are constant. The load, however, can vary considerably. A Stirling engine and its load is a dynamic system and any unbalance between its input power and output power in excess of power losses will disturb its existing equilibrium state. Thus, the stability of a Stirling engine depends on both the engine and the load it drives. Engine dynamics following a disturbance such as a load change may result in an unsatisfactory overall system performance unless effective controls are introduced into the system. A control system has to be fast enough to respond to the disturbance to avoid engine damage due to overheating or mechanical stress.

Vibrating systems can be either linear or non-linear systems. In a linear system the superposition principle applies. For example if the periodic excitation force or the load applied to a system doubles the response of the system also doubles in a linear fashion. In a non-linear system the superposition principle does not apply. It may be that the response depends on both the frequency and the amplitude of the excitation or upon other system parameters such as non-uniform temperature or pressure distribution. Another characteristic feature of a linear system is that it has a singular position of equilibrium. Any small change of operating conditions will either result in a change of equilibrium or lead to an unstable state. Non-linear systems can have more than one equilibium position depending on the conditions of equilibrium. Analysis of non-linear systems is very difficult and so often systems are linearized to simplify analysis. Unfortunately, some phenomena, especially stability, may not be properly predicted by a linear analysis.

In actuality if most systems were totally linear instabilities would be common as a result of changing conditions. It is the non-linear nature of restoring and damping forces that prevent run away conditions. The FPSE is no exception. Sufficient non-linearity must be available to ensure stability. Even though the system behaves linearly at any operating point, its stability is only possible because of the nonlinearities of the internal processes or the load.

Damping forces or losses play an important role in the FPSE, especially due to the unconstrained dynamic nature of the piston and displacer. Damping forces include a variety of internal mechanical and aerodynamic frictional forces as well as the resistance to motion imposed by loading devices driven by the engine. The internal damping forces profoundly influence the dynamic characteristics of displacer motion, while the load resistance primarily influences the piston dynamics. Of course, the piston and the displacer are closely coupled through gas pressure and thus in the FPSE the engine and load dynamics can not be separated.

Damping is a force which resists motion. Viscous damping where the resisting force is proportional to velocity, is the easiest to describe mathematically. Many practical damping devices or dashpots are of the type where a fluid, liquid or gas, is squeezed through an orifice to create a resisting force. The damping force is then proportional to the square of the velocity of relative motion in the damper. The frictional drag (or Coulomb damping) of dry sliding surfaces is virtually independent of velocity and exerts a nearly constant drag opposing the motion of a vibrating mass. Hysteresis damping is nearly always present in vibrating systems with elastic restoring forces although often it is insignificant compared to other loss mechanisms. It arises because of internal friction effects in a mechanical or gas spring system experiencing repeated cyclic flexing. In the FPSE, however, the gas springs experience a gas hysteresis loss which can result in an appreciable consumption of energy.

1.3 FPSE Models

It is necessary to model the dynamics of a free-piston Stirling engine:

- for design purposes
- to predict operating characteristics, and
- to predict and provide stability

Early Stirling engine models were simple linear models (known as 1st and 2nd order) and were used for design purposes and in an attempt to predict operating characteristics. However, they often were inaccurate in predicting losses and efficiency and typically did not attempt to predict stability, especially for the "free-piston" Stirling engines. They often used "calibration factors" or empirical correction factors gathered for a specific machine design to account for losses. As the complexity and accuracy of models increases an attempt is being made to properly represent the restoring and damping forces. The next section of this report (Chapter 2) reviews common models which have been developed for FPSE's. Then the loss mechanisms which occur in such engines are discussed (Chapter 3).

Use of accurate damping terms was limited in early FPSE models for two reasons. First, accurate representation of the terms usually required nonlinear representation resulting in complex programs requiring extensive computer time. Second, the physical mechanism of the

loss was not fully understood, often because the phenomena was difficult to independently instrument and study. Over the last few years computer capabilities have advanced in conjunction with increased understanding of the loss mechanisms in FPSE's. Models progressively account for more accurate damping terms. Chapter 4 reviews the method of accounting for damping terms in several recent FPSE models.

2. FREE-PISTON STIRLING ENGINE MODELS

2.1 Introduction

The first in-depth analysis of the Stirling cycle dates from the work of Gustav Schmidt in 1871. Since then many Stirling engine models have been developed, most of the work having been performed since the 1960's. Since the focus of this report is on the free-piston Stirling engine most of the models discussed below were developed for the FPSE. A few other Stirling modeling efforts are presented, however, primarily when the loss mechanisms included are of interest. This presentation makes no effort at reviewing all kinematic Stirling engine models, but will attempt to include all well known free-piston Stirling engine models.

A typical treatment of Stirling engine models is a chronological review of the development, introducing a person or group, their techniques, and how they were applying their models. A number of such reviews exist and the reader is refered to them for further information [7]. This brief review will take a different approach. It has been the practice to classify models into three categories: first, second or third order models. Typically as the complexity of the model increases the type and degree of losses accounted for increases. Since the primary emphasis of this report is on loss mechanisms models will be presented in order of increasing complexity according to the 1st, 2nd, and 3rd order classification. The criteria used for this classification rely on the assumptions used for modeling the engine and the complexity of the mathematical solution.

First order models are the simplest, based on the concept of an ideal-cycle analysis and using empirical parameters for a specific engine. Such models typically result in explicit equations for the overall performance parameters (such as frequency and power out). These models are primarily used for design purposes and do not take into account losses or non-linearities.

Second order models use more detailed modelling equations but still do not consider losses within the dynamic analysis. Loss terms are often included in the power calculations after the force and energy calculations are completed and thus are known as "decoupled". The assumption is that their effect is additive.

Third order models attempt to fully describe the thermodynamic process undergone by the working fluid at each instant of time and at each position within the engine. Losses are included in varying degrees throughout the calculations and one or more may include nonlinear effects. Typically this approach results in a set of simultaneous differential equations requiring time-consuming numerical solutions. The conservation equations are applied to representative control volumes (nodes or finite cells) within the engine. Thus, this method is often known as a nodal model.

2.2 Stirling Engine Model Techniques

first order

First order models utilize limited information about a specific engine to estimate basic engine parameters such as efficiency and output power. Typically input data utilizes little more than the heater and cooler temperatures, compression and expansion volumes, and frequency. The original first order model developed by Schmidt assumes that the compression and expansion spaces are maintained at isothermal conditions. Other assumptions include the working fluid is an ideal gas with a total constant mass (no leakage) and the instantaneous gas pressure is constant throughout the volumes. The resulting set of equations can be explicitly integrated and for sinusoidal volume variations closed-form solutions to the integrals can be obtained. For this model all losses are ignored.

In real machines the working spaces will tend to be adiabatic rather than isothermal. In addition in the first order model no information is available about the variables (such as pressure and temperature) throughout the cycle or as a function of position within the volumes.

First order models are used for preliminary system studies to evaluate how engines perform. For a FPSE a first order model can be used to approximate the frequency and phase angle. In general they are not useful for accurate determination of operational characteristics (such as power) or for prediction of stability.

second order

Second order models again use a relatively simple Stirling engine analysis to calculate approximate power output, heat input, and frequency. Thermal and dynamic losses are calculated by simple methods or from empirical measurements and are added to the required heat input or deducted from the output power. Typically the losses are assumed to be constant (i.e., throughout the cycle) and do not interact with each other or with the dynamics of the engine. Thus, second order models are sometimes known as "decoupled". The basic model calculations using the Schmidt analysis are still relatively simple, although some models improve upon reality by assuming adiabatic conditions [Fin, 1960, 1961]. Complexity is added depending on how accurately one attempts to develop the loss terms to be used afterwards. The various losses accounted for are the same ones as incorporated into the third order models and will be discussed in more detail in the next Chapter. The value of the power calculated by the second order model tends to be more accurate than the first order models, but second order models are still inadequate to account for nonlinearities, loss interaction, or to predict stability. Examples of second order isothermal models include Philips [], Berchowitz [55], Finkelstein [], MTI [], ORNL, and Martini [], and a second order adiabatic model by Rios [].

third order

Third order methods further increase the accuracy and complexity of Stirling engine models. A number of references have reviewed various third order models [15, 51, 53, 52, 56, Urieli]. Kankam and Rauch review and contrast FPSE third order models, especially with respect to stability analysis and incorporation of losses.

Third order models typically break the engine working spaces into a set of nodes each of which has a set of differential conservation equations (energy, mass, and momentum). The complete set of differential equations throughout the engine is solved simultaneously. Short time steps are required for a stable solution requiring a large computing effort to solve numerically. The models become even more complex as various losses are added and interactions between losses are considered.

The resulting set of equations can become unmanageably complex and various assumptions and simplifications are often made. The more important ones include:

- working space pressures, and the piston and displacer motions are sinusoidal
- isothermal conditions (some models allow for adiabatic representation
- linearization of spring force constants, and damping constants (at least near equilibrium)
- gas pressures uniform within a region (expansion, compression, heater, cooler, regenerator)

As opposed to the strict numerical technique a couple of efforts utilize a different nonnumerical approach called linearized harmonic analysis [Rauch see 15p58, 14, 15]. In this approach the variables are represented by harmonic functions. Nonlinear terms are represented by fourier series, which are truncated to retain only the sine, cosine, and constant terms. This representation allows a fairly realistic treatment of the nonlinear terms representing various losses.

2.3 Specific Models

2.5.1 Benvenuto, Farina, Monte

Benvenuto, Farina, and Monte develop a dynamic model using two equations of motion (eq 1, p 66) similar to Redlich and Berchowitz, and two thermodynamic models (both

an isothermal model and an adiabatic model) to obtain the energy exchanges in the different machine parts. The thermodynamic damping matrix, $[C_T]$, is shown to be a non-linear function of the piston and displacer velocities and includes both direct and coupling terms. The authors linearize the working gas pressure, pressures in the gas springs, and pressure drop in the heat exchangers (following Berchowitz) to obtain a set of linear, homogeneous equations of motion with constant coefficients. The set of equations is solved using Laplace transform techniques and a stability criterion is developed which takes into account the various losses. The authors also develop a technique to study the effects of variations of the load on stability using the isothermal thermodynamic model.[41]

2.5.2 Berchowitz/Sunpower

Redlich and Berchowitz develop linearized equations of motion for the piston and displacer using a Taylor series expansion about a steady-state operating point.

$$\begin{bmatrix} m_d & 0 \\ 0 & m_p \end{bmatrix}\begin{bmatrix} \ddot{x}_d \\ \ddot{x}_p \end{bmatrix} + \begin{bmatrix} D_d & 0 \\ 0 & D_p \end{bmatrix}\begin{bmatrix} \dot{x}_d \\ \dot{x}_p \end{bmatrix} + \begin{bmatrix} K_d & \alpha_P \\ \alpha_T & K_p \end{bmatrix}\begin{bmatrix} x_d \\ x_p \end{bmatrix} = \begin{bmatrix} 0 \\ 0 \end{bmatrix}$$

where m_d and m_p are the displacer and piston masses

x_d and x_p are the displacer and piston positions

D_d represents the damping on the displacer due to various losses

D_p includes similar damping plus the load on the piston

K_d and K_p represent the spring forces acting on the displacer and piston, and can be determined from engine geometry, pressures, and temperatures

α_T is the coefficient of the thermodynamic effect of the piston on the displacer, and

α_p is the coefficient of the thermodynamic effect of the displacer on the piston.

Schmidt thermodynamics is incorporated into the equations (K_d, K_p, α_T, and α_p), where α_T and α_p are approximated using a simple thermodynamic analysis. The approach is then to take the Laplace transform of this set of thermodynamic equations and plot the roots of the characteristic polynomial in the complex plane. By using Nyquist criterion on the characteristic equation they obtain necessary conditions for system stability, including the development of criteria for engine startup and stability of piston and displacer resonances.

2.5.3 Das and Bahrami

An early paper on the stability of FPSE's was presented by Das and Bahrami [43] in 1979. The primary purpose of the paper is to discuss the issue of control for Stirling engines. They present a cursory outline of non-coupled equations of motion for the displacer and the piston they do not provide any details on the coefficients or solution techniques. They assume a sinusoidal forcing function and obtain the displacer and piston equations of motion. Using Laplace transform techniques and Routh's stability criterion they discuss stability and control of Stirling engines. They refer to their model as a second order model and use it to recommend a control method by regulating the working gas pressure.

2.5.4 Gedeon

Gedeon reported on a computer model developed to utilize a personal computer [18, 20]. In order to reduce computation requirements he developed an alternate approach to the classical third order numerical node technique. He refers to such classical methods as "time-marching" methods, refering to the solution of a set of time-dependant conservation equations at each point in a mesh and continuing to solve these equations for small time-steps as time progresses. His is a globally-implicit, finite-difference method used to solve the gas dynamic equations simultaneously over the full cycle. These finite-difference equations on the entire

computational grid are solved simultaneously (three or four equations for each of over 75 nodes). In the initial presentation of Gedeon's model (GLIMPS) predicts friction losses in a preprocessor [20] and has provisions for a number of other losses in a postprocessor. This implies that the code treats loss interactions and effects in a second order model approach by accounting for additive effects on the overall power and efficiency after the dynamic motions are determined. GLIMPS is not designed for the study of FPSE, since it relies on constrained motion. Nevertheless, it was presented here, despite its lack of accurate loss models for its ability and potential to be a widely available and portable code.

2.5.5 Lazarides, Rallis

Another effort at developing Stirling models for use on a desktop computer was presented by Lazarides and Rallis [25]. Their goal is not accurate Stirling engine analysis, but to develop simple codes for the use in the design of Stirling engines. For this reason most of their effort is directed toward first and second order models. They do intend to develop a number of modules (or subroutines) to calculate a variety of losses. These codes can then be used for design optimization. Details of the code implementation were not included in the references. It is merely included here as another future potential tool for the easy calculation of losses. They do not specifically address free-piston Stirling engines and are not concerned with engine stability.

2.5.6 LeRC

A Stirling engine model was developed by Tew, Jefferies, and Miao [9,8] at the NASA LeRC. Initially the model was developed as a design optimization and performance prediction code for kinematic engines. Thermodynamics were modeled in each control volume using conservation of energy, conservation of mass, and equation of state. The model utilized the speed, mean pressure, and fixed heater and cooler temperatures measured for a specific kinematic engine to calculate power and efficiency. Later it was modified in the areas of engine thermodynamics, piston/displacer dynamics, and engine geometry to predict the performance of a free-piston Stirling engine [see 6,ref2]. The model makes simplifying assumptions that gas inertia and pressure-wave dynamics can be neglected in predicting Stirling engine performance.

The model was later modified to incorporate a number of loss mechanisms [6] to decrease the calibration effort which had been previously used in order to get the model predictions to agree with actual engine data. Calibration parameters based on engine test data had been used to bring predictions and experimental data into agreement. These parameters were a set of multiplication factors and coefficients used to adjust predicted pressure drops, heat transfer, and gas flow rates so that the code predictions could better agree with a specific engine's test data.

The nature of this model make it a very engine-specific code. In order to utilize the code for another engine design it would have to be calibrated for that engine.

2.5.7 MTI

Rauch has developed Stirling engine models for qualitative applications in design and performance evaluation [10, 11, 13]. The codes (including HSCAC and HFAST) are based on the solution of a one dimensional set of equations including continuity, energy conservation, and momentum conservation. He assumes that the solution of the governing conservation equations are harmonic functions of time. The solution is then found by solving a system of nonlinear, algebraic equations instead of a set of differential equations. The equations are linearized. Other simplifying assumptions include:

- component dynamics are not coupled to the thermodynamics
- working gas is assumed to act as a linear spring
- dissipative effects of the load are assumed to be linear dampers
- other losses considered are linear and can be represented constant damping coefficients

Using Gaussian elimination this method yields the steady-state solutions directly giving the piston and displacer motions and the phase angle at the assumed frequency. The model is designed to provide quick, reasonably accurate solutions, but is not adequate for detailed stability analysis for FPSEs.

2.5.8 ORNL

A simple thermodynamic model for Stirling machine performance was developed using a linear harmonic analysis approach instead of the numerical solution to sets of differential equations [14, 15, 16]. By representing variables in terms of harmonic oscillations and representing the nonharmonic terms in the conservation equations with truncated Fourier series the equations can be solved in a semi-closed form, resulting in a more intuitive understanding of Stirling engine behavior. In addition, the theory includes a Second Law analysis so that the efficiency and power losses resulting from effects of adiabatic cylinders, transient heat transfer, pressure drop, and seal leakage can be accounted for simultaneously so that the degree of loss coupling can be assessed.

Thus, this technique is based on three important assumptions:

(1) linearization of the ideal gas law

(2) representation of all time-dependent variables with harmonic functions

(3) replacement of terms in the governing equations that contain products of harmonic functions with truncated Fourier series.

These assumptions reduce the nonlinear differential equations to a system of almost linear algebraic equations that are solved using standard matrix algebra, with iterations as required.

Conclusions from application of this model include:

(1) nonisothermal cylinders result in both mixing and external heat-transfer irreversibilities, but mixing loss is the smaller by an order of magnitude.

(2) transient heat-transfer loss is zero for both adiabatic and isothermal cylinders and reaches a maximum for intermediate cases.

(3) the coupling between pressure drop (flow loss) and mass leakage losses can be significant.

An important advantage for the study of free-piston Stirling engines is that in the linear harmonic analysis method, the thermodynamic losses, as well as their interactions, are included intrinsically in the FPSE dynamic solution. Sample calculations using the program have shown that substantial errors in predicting dynamic behavior can occur if an isothermal calculation is used to represent an adiabatic FPSE. Linear harmonic analysis predictions have also indicated that unrealistic assumptions about the pressure drop losses can lead to additional errors.

The authors show with sensitivity studies using the linear harmonic analysis method that the thermodynamic loss assumptions used in an analysis can have a significant impact on the predicted dynamic behavior of a free-piston Stirling engine.

2.5.9 Weiss, Walker, Fauvel

This model represents another attempt to develop a simple available model which can be used on a desktop computer [58, 24]. The authors use a second order technique based on Martini [1978, 53]. Isothermal cycles are assumed with decoupled corrections for the various losses. Although this model might be useful as a kinematic Stirling engine design verification tool due to its ease of use it does not handle the dynamics of an FPSE and losses are decoupled from the thermodynamic calculations. Loss considerations were not discussed.

3. LOSSES IN THE FREE-PISTON STIRLING ENGINE

3.1 Introduction

Understanding [51, 55] the losses in a free-piston Stirling engine is important to accurately predict output powers and efficiency, for optimum design, and to understand and predict stability. Indeed, the losses are critical factors in establishing and maintaining engine stability under varying load and input conditions. In the first order model losses are ignored and only rough estimates of the power output are obtained. In the second order model approximate average losses are calculated to improve upon the power calculation. Calibration factors are calculated which are used to adjust the second order model calculations to match actual engine performance.[6] Finally, third order models attempt to incorporate losses to varying degrees, including their nonlinear nature and interaction with each other into the equations of motion. This section will review the loss mechanisms most often considered in the development of Stirling models, including current research efforts to understand and develop loss models.

Losses in free-piston Stirling engines can be divided into two categories: thermal losses and dynamic losses. Thermal losses result primarily from the conduction of energy throughout the structure of the engine. They represent a relatively simple loss of energy proportional to the temperature differences throughout the various solid and gaseous components. The net effect of this thermal loss is a decrease in thermal efficiency, requiring more energy input to achieve the desired work output. They have little effect on the dynamics of the FPSE and will be discussed only briefly.

Dynamic losses, on the other hand, are more complex to predict and can be a function of oscillator frequency, gas pressures, temperatures, and displacer/piston position or velocity. They can be nonlinear and have been shown to interact (effect each other). They provide the limiting factor which determines stability of the engine to varying conditions. These are the losses that must be incorporated into an accurate third order model that attempts to predict stability. Dynamic losses include viscous dissipation, appendix gap, leakage, and gas spring hysteresis losses.

It is very difficult to properly instrument an FPSE in order to obtain the measurements needed to understand and characterize these losses. In order to describe the physics of the losses and develop accurate models information is needed on the (1) temperature distribution in the working space and metal walls, (2) flow and pressure distributions in the working spaces, and (3) leakage paths and flow rates to and from the working space.

- Thermal losses
    - conduction losses
    - cylinder heat transfers
    - regenerator enthalpy loss
    - external radiation and convection
- Thermodynamic losses
    - mechanical friction
    - aerodynamic friction (fluid viscous losses)
    - gas spring hysteresis losses
    - seal leakage losses
    - heat transfer between gas and walls
    - mixing losses
    - appendix gap losses
        - shuttle heat transfer
        - gas enthalpy transfer
        - hysteresis heat transfer
    - adiabatic working space loss

Tew has contributed several reviews of current loss studies at different institutions [1,5,7]. His summaries as well as references to each groups are incorporated into the following brief review.

3.2 Thermal losses

In an ideal Stirling engine all of the thermal energy goes to gas expansion. Any thermal energy not converted to mechanical work would be rejected in the cooler. In a real Stirling engine, however, a number of losses rob the system of some of the original thermal energy before, during, and after conversion to mechanical energy. An important loss of thermal energy that occurs before gas expansion is the conduction through the solid engine materials and through the gas, in effect shunting the desired path for energy flow. These conductive losses occur regardless of the motion of the engine components and have little effect on the engine dynamics. Their principle effect is in the thermal efficiency of the engine. Conductive losses occur throughout the engine wherever there is a temperature difference, but are most significant in materials with a large temperature gradient. Thus, for instance, the cooler and heater have little conductive losses while the regenerator can have a significant conductive loss, with the piston and displacer conductive losses to a lesser degree.

In addition to simple conduction losses there are several other thermal losses [7,54, 51, 55] included in a list of thermal losses:

- conduction through engine solid materials (e.g., regenerator, piston, and displacer)
- shuttle loss (also discussed in appendix gap losses)
- convection at exterior engine surfaces
- radiation from exterior engine surfaces

The conductive losses are the simplest to deal with in FPSE models. Knowledge of the engine geometry, materials used, and temperature distributions allow basic conduction calculations. Since conduction losses have little effect on engine dynamics and are primarily used in the models for the determination of efficiency they will not be discussed further.

4. DAMPING LOSSES USED IN FREE-PISTON STIRLING ENGINE MODELS

4.1 Benvenuto, Farina, Monte

In this model the authors use damping coefficients in the two equations of motion [37, 39, 41]. These damping coefficients incorporate losses due to:

(1) pressure drop resulting from gas flow through the regenerator *{uses a simples flow model assuming laminar flow [39]}*
(2) pressure drops resulting from gas flow through the heat exchangers *{ uses a simple flow model assuming turbulent flow [39]}*
(3) effect of external load
(4) hysteresis *{uses model from Urieli/Berchowitz [54]}*

The authors do incorporate in their model a coupling term (hysteresis loss) between the displacer and the piston.

The authors discuss the stabilizing effect of pressure losses [39] and incorporate damping and spring coefficients into the stability criteria. In addition the authors later examine the combined effects of load variations and losses on stability.

4.2 Berchowitz/Sunpower

The primary purpose of this reference is to examine techniques for stability analysis. Development of the two thermodynamic equations of motion are outlined, but details of coefficients, etc. are not provided. The following information on losses included in the model is given:

- the damping term (D_d) in the displacer equation is due primarily to "viscous forces on the moving gas", or flow losses, presumably in the regenerator and heat exchangers.
- the damping term (D_p)in the piston equation includes both similar flow losses and the load
- other "incidental" irreversible losses, such as gas spring hysteresis have been partly accounted for in D_p and D_d; development or values of the damping terms are **not** included
- coupling of the displacer to the piston is alluded to but not included, thus the damping matrix contains only diagonal terms and no coupling terms. The authors state that coupling terms could be included by "staightforward extension of the analysis".

Insight into how the authors may have used loss terms in their model may be obtained by examining a later publication by one of the authors [55]. In his thesis [55] Berchowitz presents details on the development of models for damping terms for Stirling engines (note that he even uses the same notation as ref. 42). Some of the highlights of his work include (page numbers refer to ref 55):

Flow losses (pp 111-124, 145, 185-190)

• assuming turbulent flow in the heat exchangers he develops a nonlinear expression for the pressure drop across the exchanger; then linearizes expression

• assuming turbulent flow in the regenerator he develops an approximately linear expression for the pressure drop

• from these pressure drops he develops expressions for Dd and Ddp (the coupling effect of the piston on the displacer)

• shows that these damping terms increase with increasing displacer and piston amplitudes, stabilizing oscillations

• illustrates that if the flow through these components is entirely laminar damping is not stabilizing

Hysteresis loss (pp 212-225)

• develops an approximate model for hysteresis loss using a somewhat simplified theory

• a number of assumptions are made, one of which is that coupling to other loss mechanisms (viscous dissipation, leakage,etc.) is ignored

• gas spring hysteresis is a function of wall temperature, volume ratio, wetted area (area of contact between surface involved and gas), gas type, and various material properties (thermal conductivity, thermal diffusivity, and heat capacities)

• this development can not be applied to the gas springs but also to the gas in the expansion and compression spaces

• hysteresis loss may account for as much as 10 % of the lost energy

Seal leakage (pp 226-241)

• details the development of an expression of mass flow for a clearance type seal (the type used in gas supported FPSE's)

• seal leakage not only differentially pumps, but also results in some viscous dissipation

• the effect of seal leakage on piston power and engine dynamics can be accomplished by modifying f, the phase angle between the piston and the displacer, by an equation developed

• leakage loss is a function of the clearance gap, pressure amplitude, and the eccentricity of the two parts moving with respect to each other

• free-piston engines are particularly sensitive to seal leakage, since any differential in mean pressures over any of the moving parts will cause them to drift. The solution to this problem is to provide an equalizing port

• leakage could account for as much as 10 % of the lost energy

Other incidental losses (pp 241-281)

• Berchowitz also discusses several other losses (appendix gap losses and mechanical losses), but it is unlikely that they were included in the FPSE model [42].

4.3 Das and Bahrami

This model does not give any details on spring or damping constants, nor does it discuss or mention any type or accounting for losses. It is assumed that they use details from other second and third order models and focus on stability and control analysis. Thus, it is likely that this model and specifically this reference has nothing to contribute to the incorporation of losses in FPSE models.

4.4 Gedeon

The model developed by Gedeon for use on a desktop computer accounts for "friction" losses in a preprocessor. This implies that simple representations for mechanical and viscous damping are accounted for (probably subtracted from the energy input) prior to solving the set of equations. The author implies that the code includes flow losses and mixing losses, but they are most likely not interactive, since they are calculated in the preprocessor.

Similarly a postprocessor has provisions for the following losses:

- gas leakage
- shuttle transfer loss
- gas spring hysteresis loss
- appendix gap losses
- other thermal losses

Similar to "calibration" or correction factors used in second order models these postprocessor provisions must subtract power from the output to predict operational characteristics. GLIMPS is not designed for the study of FPSE, since it relies on constrained motion. Thus, it is not adequate for the study of stability, especially in FPSE's.

4.5 Lazarides, Rallis

These authors may in the future provide simple modules or subprograms for the design and optimization of Stirling engine components and systems. Models for loss calculations may be included.

4.6 LeRC

This model incorporates the following loss considerations into its engine-specific performance calculations:

(1) assumes completely laminar, steady flow in heat exchangers and regenerator, uses heat-transfer and friction-factor correlations for the specific tube cross-sections to estimate viscous flow damping
(2) displacer appendix gap pumping loss equation is used
(3) gas spring is represented by a spring constant and a linear damping factor
(4) gas leakage is estimated on specific geometry
(5) models the shuttle loss

The model does not consider:
- gas spring hysteresis
- non-steady-state conditions in the heat exchangers
- leakage at the centering port
- interaction among losses

4.7 MTI

In the HSCAC computer simulation code [11] "flow friction" losses are included directly in the equations of momentum and energy. Several additional "parasitic" losses are also considered by the code, including:
- piston seal leakage out of the working space
- displacer seal leakage between the expansion and compression spaces
- thermal hysteresis in the expansion and compression spaces
- shuttle conduction along the displacer wall

Details on the development or value of these losses is not provided.

In the HFAST code [13] flow friction is developed in terms of conventional friction factor, entrance and exit loss coefficients, and an average velocity head. Thus the code may not accurately represent or recognize some of the losses, heat-transfer, and aerodynamic phenomena associated with oscillating flow. In addition the codes assume that coupling effects between losses are negligible.